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CARBON-PHENOLIC CAGES FOR HIGH-SPEED BEARINGS

Part II - Bearing Evaluation with a Multiply-Alkylated Cyclopentane (MAC) Lubricant

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THIS TECHNICAL REPORT HAS BEEN REVIEWED AND IS APPROVED FOR PUBLICATION.

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14. ABSTRACT

This is the second part of a three-part series of reports to investigate carbon-phenolic bearing cages in high-speed, lightly lubricated bearings. This portion covers full-scale bearing testing with carbon-phenolic and cotton-phenolic cages impregnated with multi-ply, alkylated, cyclopentane (MAC) lubricant, commercially known as Pennzane®. Experimentally, it was found that bearings fitted with the carbon-phenolic cages generate more heat than bearings fitted with the cotton-phenolic cage. This is the opposite of the intended result. The difference is attributed to damage of the steel raceway with carbon-phenolic cages. This occurred because of wear debris generated from the carbon fibers in the ball pocket. Additionally, the study provides a useful assessment of heat generation in lightly lubricated bearings with composite cages and the effect of thermal preloading on bearing heat generation.

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Table of Contents

List of Figures

List of Tables

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1. Introduction

The objective of this effort was to develop a new bearing cage material made from a carbon fiber-phenolic resin matrix (carbon-phenolic), with the end goal of producing a material that would have better performance than cotton-phenolic cages in high-speed, lightly lubricated bearings. This is the second part of a three-part series of reports. This report addresses the experimental bearing evaluation. Part I addresses the initial material selection, mechanical and thermal characterization, and tribology testing, and Part III covers thermal modeling of the bearing. There is also a separate set of reports in progress from the Air Force Research Laboratory Materials Directorate that addresses testing in vacuum and hard coatings on the bearing steel.

The rationale for selecting carbon-phenolic as a candidate material was based on the potential to significantly improve the mechanical and thermal properties of the cotton-phenolic material. Based on prior experience with carbon matrix - carbon fiber cages (C-C) [1-3], we anticipated that replacing cotton fibers with carbon fibers would greatly improve the thermal conductivity, strength, and modulus of elasticity, while also decreasing the coefficient of thermal expansion. Additionally, carbon-phenolic cages would be less expensive than C-C due to the cost associated with generating the matrix of the composite material. We also hoped to lower the coefficient of friction (COF) of the carbon-phenolic matrix by incorporating lubricants into the matrix of the cage. In Part I, we found that the mechanical and thermal properties were substantially better than cotton-phenolic, but the friction was approximately the same in both materials. We also found that there was not a substantial benefit with solid lubricants incorporated in the carbon-phenolic, but the Pennzane® lubricant did provide a substantial benefit over dry carbon-phenolic, and dry cotton-phenolic. Based on the results from Part I, the bearing testing primarily examines the carbon-phenolic and cotton-phenolic with the Pennzane® lubricant.

2. Experimental

2.1 Test Bearings

Details of the bearing used in this study are given in Table 1. The bearings came as a DF duplex set with a nominal preload of 65 lb. The bearing manufacturer was the Barden Corporation, Danbury, Connecticut. The bearings came preassembled with cotton-phenolic cages. A photograph of a typical test bearing is shown in Figure 1. A total of 15 sets of bearings were tested. A chronological listing of the bearings tested is provided in Table 2. The first column in Table 2 provides a data set name that is referred to in other figures. Each box in the table represents a different set of bearings and each row a particular test on a given day. Eight sets of bearings tested were made from 52100 steel, four sets from M50 steel, one set from T15 steel that we previously had in the laboratory, and the last two bearings tested were made from Cru 20 steel. Eleven sets had silicon nitride rolling elements, while the others had metal balls of the same material as the race material. Additional details of the material combinations are given in Table 2.

As shown in Table 2, most of the bearings were coated with the Pennzane® lubricant. As part of this process the bearings were disassembled, reassembled with the carbon-phenolic cages, and the cages were vacuum impregnated with the lubricant while assembled in the bearing. The disassembly, assembly, coating, and vacuum impregnation was performed by AFRL/MLBT.

Table 1. 206 Bearing Geometry

Class 206, ABEC 7, single outer land guided cage		Number of balls	11
Contact angle	15°	Cage OD (in)	2.049
Outer race curvature factor	0.5175	Cage ID (in)	1.800
Inner race curvature factor	0.53	Cage land clearance (in)	0.011
Pitch diameter (in)	1.81	Cage pocket clearance (in)	0.016
Ball diameter (in)	0.375	Cage width (in)	0.590
Axial preload (lb)	65	Radial load (lb)	5

Figure 1. Duplex Set of Test Bearings with Cotton-phenolic Cages

Table 2 - Chronological List of Bearing Tests Performed

NRO
AFRL/PRTM — In-House Testing Summary

Test Dataset Name	Races	Balls	Cages	Lubricant	Speeds	Comment About Test
H230	52100	52100	Cotton-phenolic	Mil 7808 squirt	10 K	New bearing 110-minute break in
H230a	52100	52100	Cotton-phenolic	Mil 7808 squirt	10 K	Nice steady state (SS) response
H230b	52100	52100	Cotton-phenolic	Mil 7808 squirt	10 K	Nice SS response
H231	52100	52100	Cotton-phenolic	Mil 7808 squirt	20 K	Ramped to 20K failed after 15 min at 20K
H232	52100	52100	Cotton-phenolic	Mil 7808 Squirt	10 K 20 K	10K break in failed after 20 min at 20K

Table 2 - Chronological List of Bearing Tests Performed (continued)

NRO AFRL/PRTM Test Dataset Name	Races	In-House Testing Summary Balls	Cages	Lubricant	Speeds	Comment About Test
H233a	52100	52100	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS
H233b	52100	52100	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS
H233c	52100	52100	Cotton-Phenolic	Pennzane® coated	20 K	Failed after 44 min at 20K
H234	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	New Bearing Two steps at 178 & 227 min
H234a	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS
H234b	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS
H234c	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	20 K	Ramped to 20 K Hi temp but no failure
H235	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	New Bearing
H235a	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS
H235b	52100	Si3N4	Cotton-Phenolic	Pennzane® coated	20 K	Failed after 26 min at 20K
H236	52100	Si3N4	Cotton-Phenolic	Durad® coated	10 K	New Bearing Try a different lube
H236a	52100	Si3N4	Cotton-Phenolic	Durad® coated	10 K	None
H236b	52100	Si3N4	Cotton-Phenolic	Durad® coated	20 K	Failed after 7 min at 20K

Table 2 - Chronological List of Bearing Tests Performed (continued)

NRO AFRL/PRTM Test Dataset Name	Races	In-House Testing Summary Balls	Cages	Lubricant	Speeds	Comment About Test
H237	52100	52100	Cotton-Phenolic	Pennzane coated	10 K	New Bearing
H237a	52100	52100	Cotton-Phenolic	Pennzane® coated	10 K	Repeat of SS for 1 h Disassembled and inspected
H238	52100	Si3N4	Cotton-Phenolic	Pennzane coated	10 K	New Bearing Ran for 5 h
H239	M50	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	New Bearing First Spring Loaded Test
H239a	M50	Si3N4	Cotton-Phenolic	Pennzane® coated	10 K	Removed Cover Plate – Ramped up to 20 K
H239b	M50	Si3N4	Cotton-Phenolic	Pennzane® Coated	10 K	30 lb spring load Ramped to 20K
H239c	M50	Si3N4	Cotton-Phenolic	Pennzane® coated	20 K	60 lbs spring load Ramped to 20K
H240	52100	52100	Cotton-Phenolic	Pennzane® coated	10 K	New Bearing 60 lb spring load
H241	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	10 K	New Bearing 60 lb
H241a	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	10K	60 lb Ramped to 20K rpm
H241b	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	10K	30 lb Ramped to 20K rpm
H241c	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	20K	60 lb Ramped to 20K rpm
H242	M50	Si3N4	Cotton-Phenolic	Pennzane® coated	10K	Cleaned & relubed bearing from H241
H242a	M50	Si3N4	Cotton-Phenolic	Pennzane® coated	20K	60 lb load Ramped to 35K rpm

Table 2 – Chronological List of Bearing Tests Performed (concluded)

H243	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	10K	Cleaned, relubed, new spring 60 lb
NRO						
AFRL/PRTM		In-House Testing Summary				
Test Dataset Name	Races	Balls	Cages	Lubricant	Speeds	Comment About Test
H243a	M50	Si3N4	Carbon-Phenolic	Pennzane® coated	20K	Ramped to 30K rpm

H244	T15	Si3N4	Carbon-Phenolic	Pennzane® coated	10K	Different Brg Vendor & mat'l 60 lb
H244a	T15	Si3N4	Carbon-Phenolic	Pennzane® coated	20K	Ramped to 25 K rpm

H245	CRU-20	Si3N4	Cotton-Phenolic	Pennzane® coated	10K	New brg mat'l 60 lb
H245a	CRU-20	Si3N4	Cotton-Phenolic	Pennzane® coated	20K	Ramped to 28K rpm

H246	CRU-20	Si3N4	Carbon-Phenolic	Pennzane® coated	10K	New brg 60 lb
H246a	CRU-20	Si3N4	Carbon-Phenolic	Pennzane® coated	20K	Ramped to 28K rpm

2.2 Experimental Test Rig

A photograph of the test rig with the duplex bearings is shown in Figure 2. In operation, a 5-inch diameter housing connected to the back plate surrounds the bearing. A cover plate is bolted to the front of the housing. The housing and cover plate are not shown in Figure 2. In some of the higher speed tests, the cover plate was removed to improve the convective heat transfer by reducing the ambient air temperature. Tests with the cover plate removed are noted in Table 2.

The test rig has been used in several studies to evaluate new concepts for cruise missile bearings [2,3] . In this effort, the rig was modified to accommodate the duplex bearings by making a new nose piece and outer race clamp to preload the bearings. In the second half of the testing, we modified the outer race clamp to use a spring to preload the bearings. The spring-loaded testing started with bearing H239 in Table 2. Most of the spring-loaded tests were run with a 60 lb spring preload, but some were run with a 30-lb spring preload. Whether the bearing test was 30 lb or 60 lb is noted in the comments in Table 2. Spring preloading was to done to ensure that the preload, on the bearings remains fairly constant regardless of thermal expansion factors. With the clamped DF bearings, the bearings rely solely on the bearing stiffness (i.e., modulus) to impose the bearing preload. A slight change in thermal gradient from the inner to outer race can have a dramatic effect on the bearing preload. The thermal effects on preload are covered in the Discussion Section and in more detail in the thermal analysis in Part III.

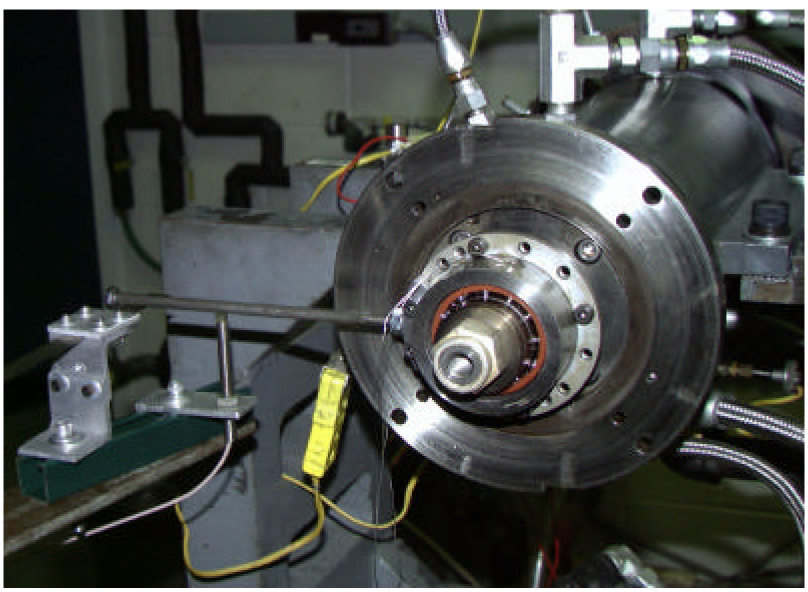

Figure 2. Photograph of Test Bearings Mounted in the Test Rig

A cross-section of the test rig is shown in Figure 3. The bearing torque was determined by the moment imposed to keep the outer race from rotating. The torque instrumentation is shown in Figure 3. Thermocouples were mounted on the outer and inner race in the locations shown in Figure 3. The bearing shaft is driven by an air turbine on the same shaft as the test bearings and approximately 18 inches away from the test bearings. This turbine gets fairly cool during operation, 10 to 0^{o}C for 10,000 to 20,000 shaft rpm. This is due to the expansion of air to drive the shaft. This provides a heat sink for the inner race, which played a role in the thermal gradients that reduce the preload.

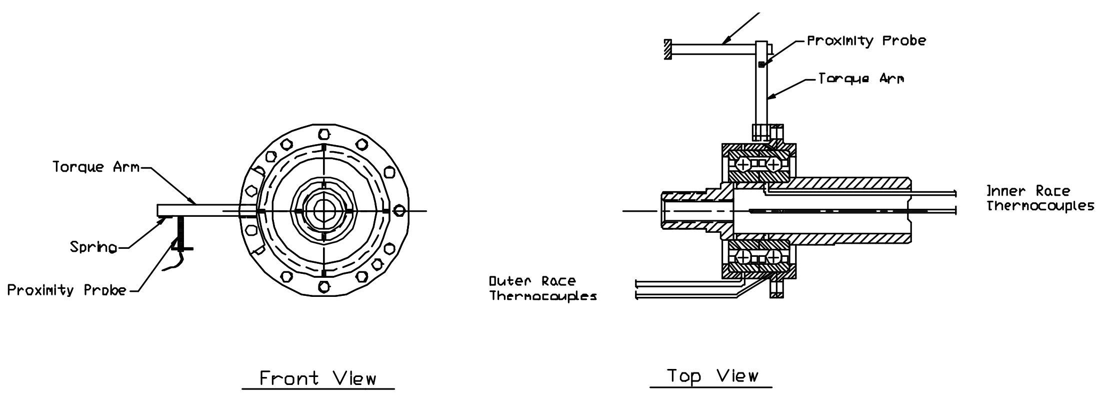

Figure 3. Schematic of Test Bearing Support, Torque Measuring Instrumentation, and Test Shaft

3. Results

3.1 Experimental Bearing Data

During the early portion of this test program, it was found that the clamped DF bearings would experience a dramatic drop in bearing temperature and torque approximately 1 to 4 hours into the testing, as shown in Figure 4. Considerable effort was spent trying to resolve if this was due to lower friction or a change in bearing preload. By the end of the program, there was evidence that both were involved.

Plots for several of the clamped DF bearings on the second day of testing are shown in Figure 5. On the second day, and subsequent days of testing, the bearings did not start with the high friction as seen in the first day. Instead, they generally approached the same steady state temperature from the day before. This indicates that the breakin of the bearings from the first day is a permanent change in friction, preload, or a combination of both. It was also found that at 10,000 rpm these bearing could be restarted on several days (up to 4 days was demonstrated) with no apparent change in bearing performance after the initial breakin period.

Other parameters shown in Figure 4 and 5, include different ball material and lubricants. Of the lubricants tested, Mil-L-7808 turbine engine lubricant produced the lowest outer race bearing temperature. Pennzane® and Durad® 620 B were similar. The Mil-L-7808 lubricant has lower viscosity than either Pennzane® or Durad® 620B at these temperatures, so this is probably a viscosity effect. Also, bearings with ceramic rolling elements ran at lower outer race temperature than bearings with metal rolling elements. This is probably an impact on friction due to asperity contact in thin elastohydrodynamic (EHD) film conditions.

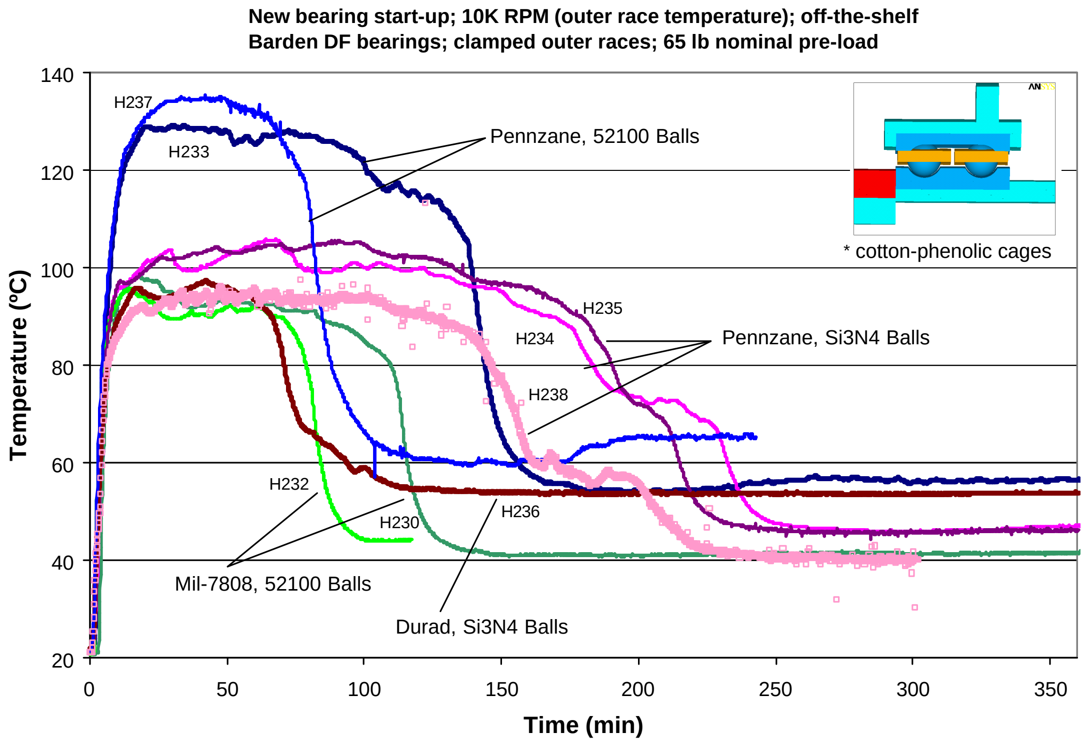

Figure 4. Plots for Bearing Outer Race Temperature during the Breakin with Clamped DF Bearings

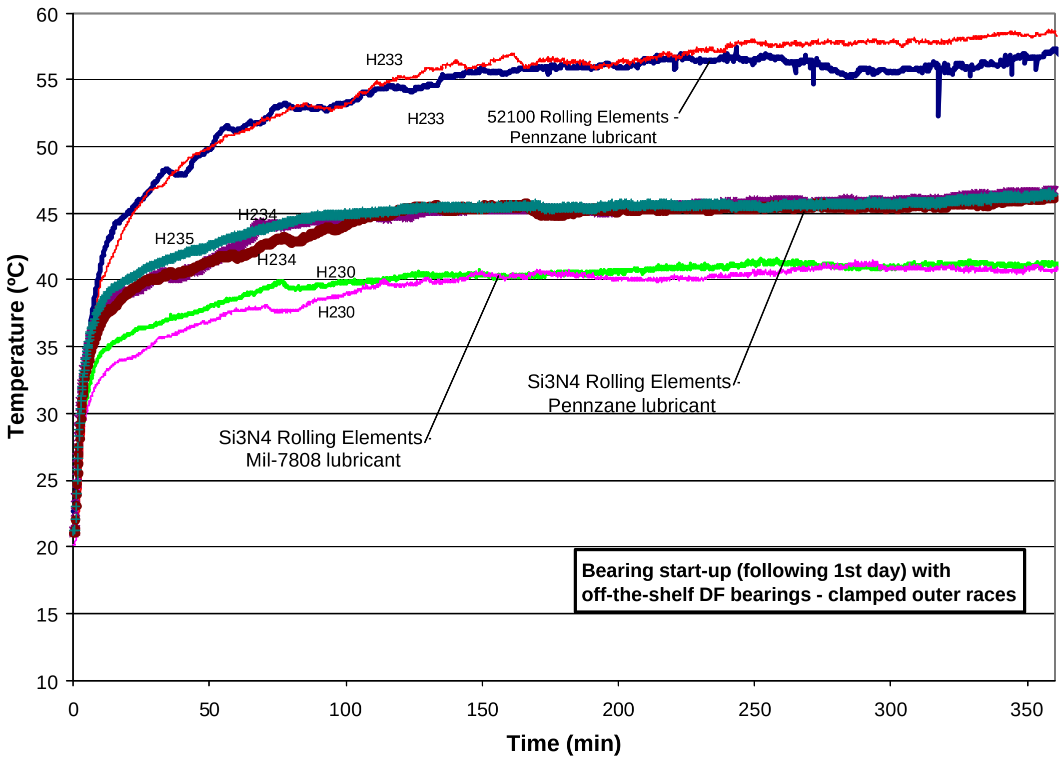

Figure 5. Plots for Bearing Outer Race Temperature on the Second Day of Testing with the Clamped DF Bearings

Several plots for the outer race temperature on the first day of testing with spring-loaded bearings are shown in Figure 6. There is still a decrease in bearing temperature with time, but the change is not as drastic as with the clamped bearings. The spring-loaded bearings are relatively insensitive to a change in loading from thermal gradients, so it is reasonable to attribute the decrease in bearing temperature in these tests to a decrease in bearing friction. Note that this decrease is not as drastic as what occurred with the clamped bearings. The decrease with clamped bearings appears to be compounded with a change in friction and preload. Figure 6 also shows bearings fitted with both cotton-phenolic and carbon-phenolic cages. In general, the cotton-phenolic cages are running about 20°C cooler than the carbon-phenolic cages. Ideally, it was intended that the carbon-phenolic would run slightly cooler than the cotton-phenolic cages.

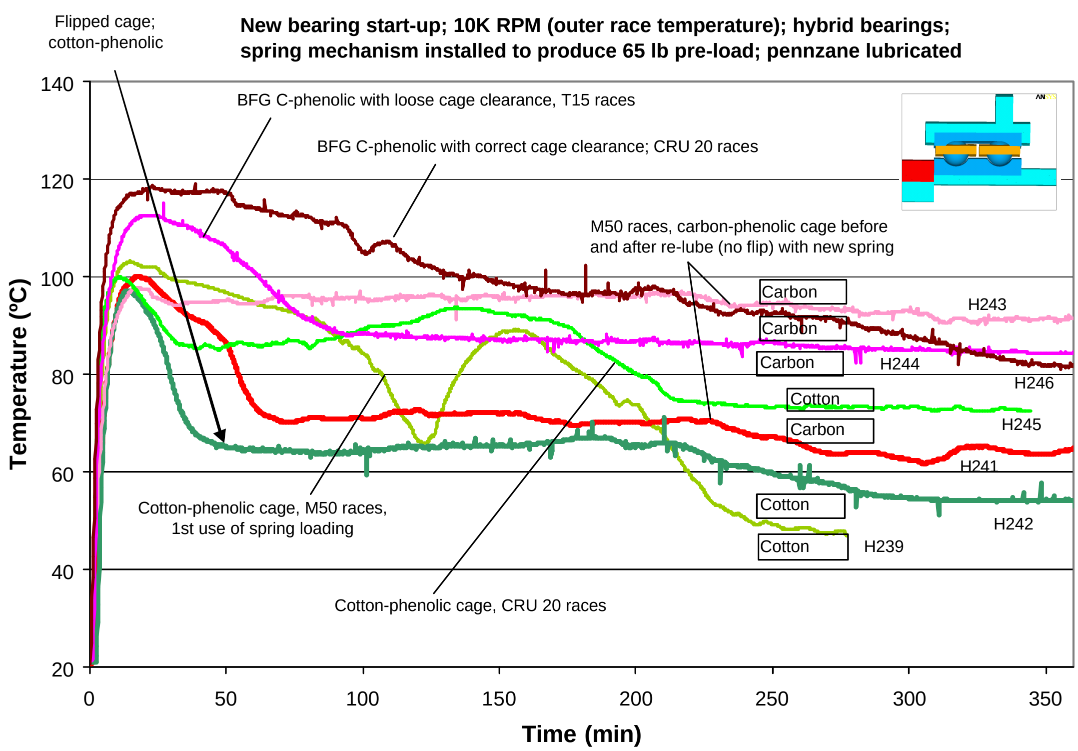

Figure 6. Plots for Bearing Outer Race Temperature for the Spring-Loaded Bearings on the First Day of Testing

Results for the last series of tests with the Cru 20 cotton-phenolic and carbon-phenolic cages are shown in Figures 7 through 9. This series of tests represent the best direct comparison of carbon-phenolic and cotton-phenolic under nearly identical test conditions. As shown in Figure 9, the bearings fitted with the carbon-phenolic cages in these tests ran about 15°C hotter. This is similar to the results previously shown in Figure 6.

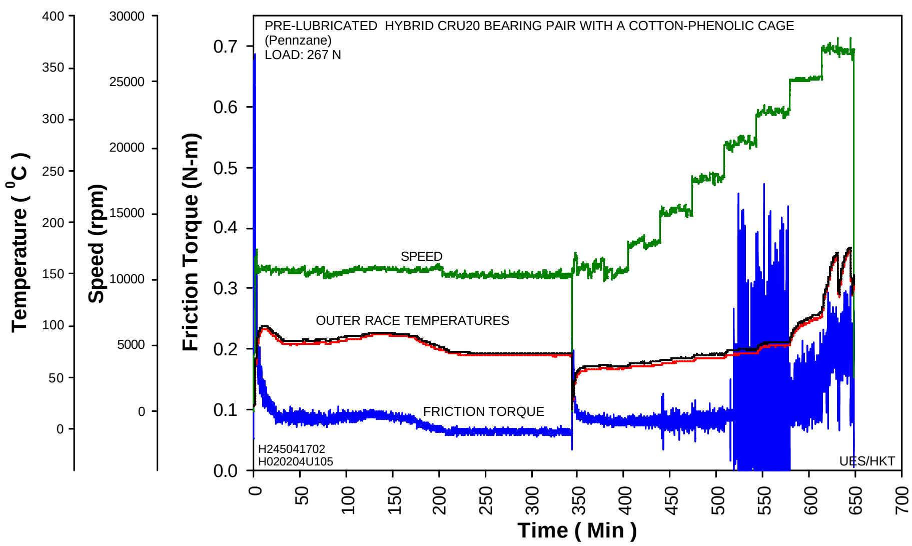

Figure 7. Outer Race Temperature and Friction Torque for the Cru 20 Bearings with the Pennzane® Lubricant and Cotton-Phenolic Cages

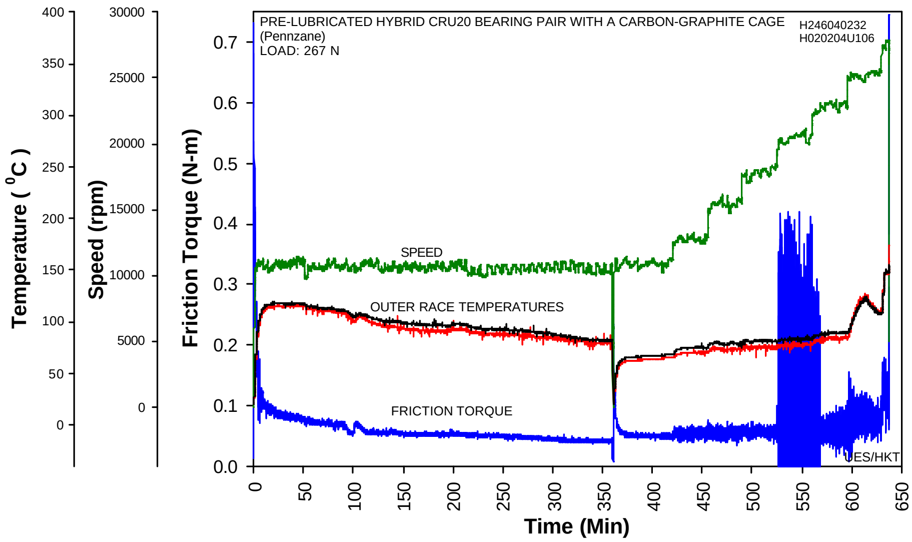

Figure 8. Outer Race Temperature and Friction Torque for the Cru 20 Bearings with the Pennzane® Lubricant and Carbon-Phenolic Cages

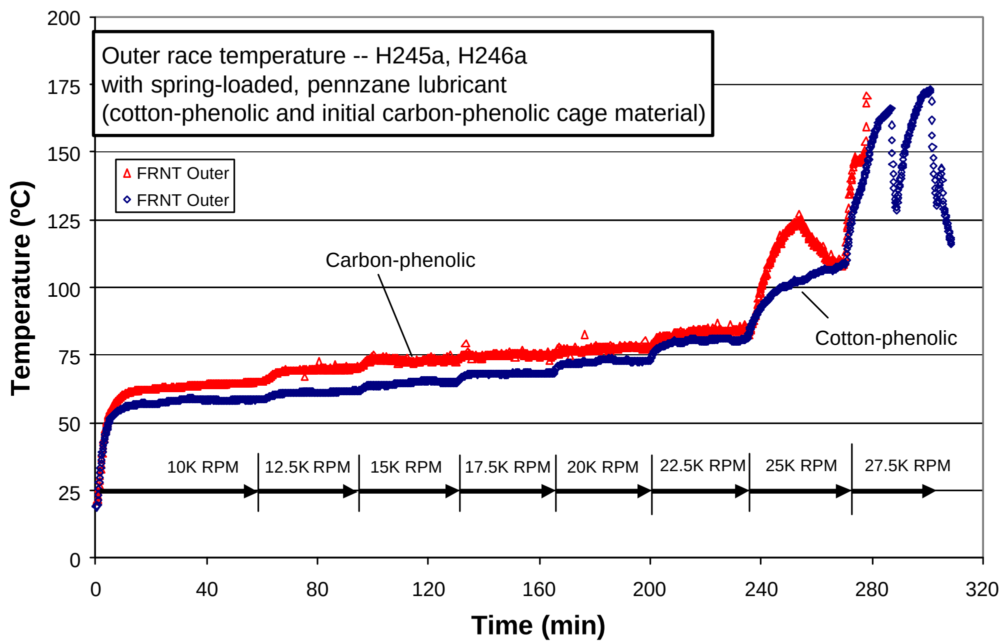

Figure 9. Comparison of Bearing Temperature with Carbon-Phenolic and Cotton-Phenolic Cages

3.2 Examination of the Bearings after Testing

The condition of the cotton-phenolic cage and carbon-phenolic cages after tests H233c and H243a are shown in Figures 10 (a) and (b). The cotton-phenolic cage is a thermal failure. The material was severely thermally degraded and apparently lost mechanical strength, resulting in destruction of the ball pocket webs. This failure occurred at 20,000 rpm. The front cover was still being used on the test rig during this time period. The bearing outer race was at a temperature of about 160^{o} C when the bearing failed. The cage in Figure 10 (b) is from a test with the cover removed but at a speed of 30,000 rpm. The cage shown in the figure was spongy, indicating degradation of the matrix. The separation of the outer layer is tearing of the carbon fibers due to centrifugal stress. A C-C composite cage, reinforced with cross-stitching, is one way to eliminate this failure mode, if needed.

Figure 10. Failure Modes of the Phenolic Cages: (a) Cotton-Phenolic Fails by Thermal Degradation of Matrix and Fibers, and (b) Carbon-Phenolic Fails by Thermal Degradation of the Matrix and Tearing of Carbon Fibers at 30,000 rpm

The condition of bearing H241 after test and prior to cleaning is shown in Figure 11. This bearing was Pennzane® lubricated and tested to speeds of 20,000 rpm, 60 lb spring load, without failure. However, there is considerable black wear debris in the bearing. This debris came from wear in the ball pockets. The land surfaces of the bearing did not have high wear. Since this wear was not seen on the land, or in the high-speed friction tests, it appears that the wear of ball pockets is aggravated by the impact loading due to ball collisions in the pocket.

Micrographs documenting the surface condition of the race and Si_3N_4 ball from bearing H241 are shown in Figure 12. The bearing has been cleaned. There is clearly surface damage in the functional areas due to rolling over the carbon fiber wear debris. The higher operating temperature in the bearing with the carbon-phenolic cages is thought be caused by this surface damage, along with graphite powder from the debris, both disrupting the EHD film.

Figure 11. Condition of a Bearing with a Carbon-Phenolic Cage after Testing at Shaft Speed of 20,000 rpm

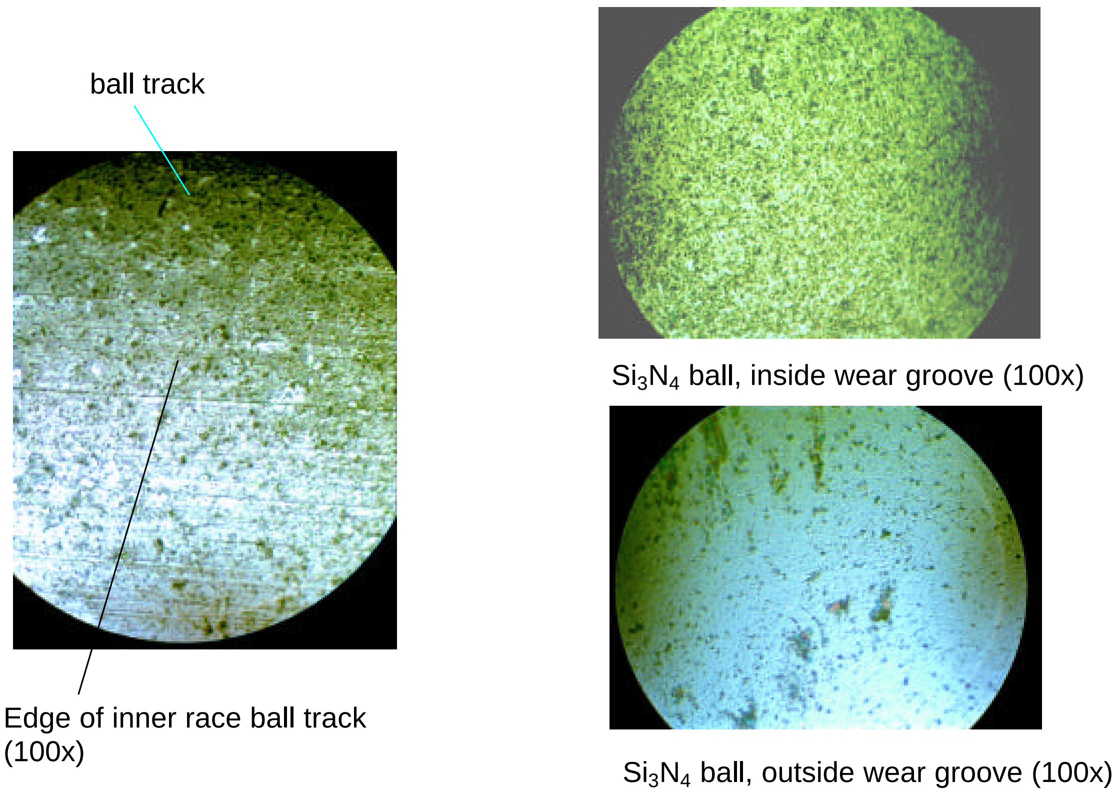

Figure 12. Micrographs of the Bearing Surface of a Bearing Tested with a Carbon-Phenolic Cage

The surface condition of bearing H239 after testing is shown in Figure 13. This bearing had a cotton-phenolic cage and tested to speeds of 20,000 rpm without failure. There is some polishing and slight oil degradation at the cage land surface; otherwise, this bearing is very clean and in very good shape.

Micrographs documenting the surface condition of the race and Si_3N_4 ball from bearing 241 are shown in Figure 14. The active surfaces of the bearing are in very good shape.

Figure 13. Condition of a Bearing with a Cotton-phenolic Cage after Testing at Shaft Speed of 20,000 rpm

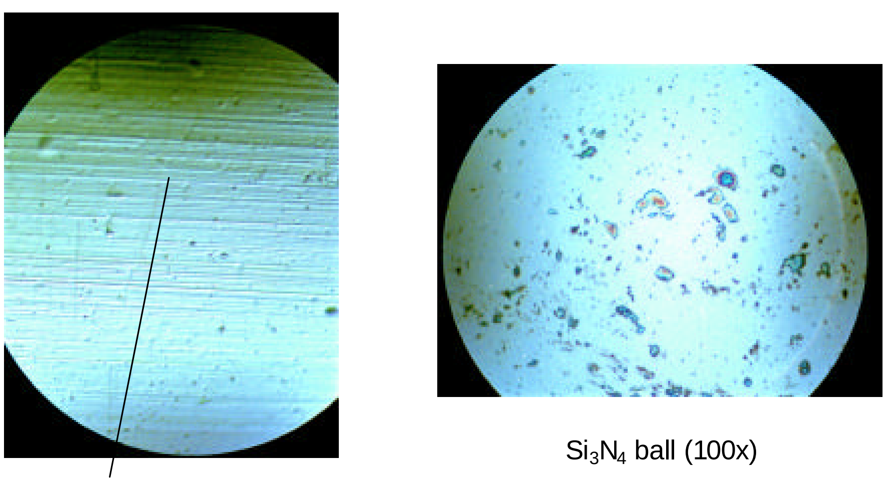

Figure 14. Micrographs of the Bearing Surface of a Bearing Tested with a Cotton-Phenolic Cage

4. Discussion

206-size bearings can operate at 10,000 shaft rpm with only a light coating of lubricant, at a moderate load of 65 lb, in air environment. As the speed increased from 10,000 to 20,000 shaft rpm, thermal management became a concern. Failure was encountered at 20,000 rpm when the environment was totally contained by a surrounding structure. By opening up the front of that structure to ambient air, we were able to achieve bearing speeds as high as 30,000 rpm. The increase in speed is attributed to improved heat transfer via convection to the ambient air. In vacuum, this form of heat transfer is not available, so considerable attention would have to be given to thermal management via structural design, heat sinks, and thermal conductivity.

The failure with the cover, or with the cover removed, was determined by bearing temperature. Essentially, when the bearing outer race temperature exceeded 160^{o} C, the bearings would fail. Failure was accompanied by dry surfaces (lack of lubricant) on the races and cage, and thermal/mechanical failure of the bearing cage. The cotton-phenolic cages experienced more distress than the carbon-phenolic cages in these failure scenarios.

The performance of the bearings was affected by preload method, clamped bearings compared to spring loaded. The bearing inner race in this particular test rig runs cooler than usual because of the air turbine heat sink at the end of the shaft. With this type of gradient outer race running hotter than the inner race, the bearing unloaded during testing. This is part of the drop in the initial breakin with the clamped bearings shown in Figure 4. With the spring-loaded bearings, there is good confidence that the load remained fairly constant at 60 lb or 30 lb, depending on the test. However, there is still a drop in bearing operating temperature with spring-loaded bearings, but not as significant as with the clamped bearing. The drop with spring-loaded bearing is attributed to a change in friction during the breakin period. The change with the clamped bearings is a combination of friction and preload.

There was a beneficial effect of Si_3N_4 rolling elements compared to steel rolling elements and Mil-L-7808 lubricant compared to the Pennzane®, in terms of reducing bearing temperature. The benefit of Si_3N_4 over steel is credited to a reduction in friction due to an improvement in boundary lubrication. The benefit with Mil-L-7808 is attributed to lower friction as a result of lower viscosity. Even though Mil-L-7808 generates lower temperatures, its higher volatility may not be attractive for a vacuum environment.

Most importantly to this program, carbon-phenolic cages did not perform better than cotton-phenolic cages in terms of bearing temperature and bearing surface condition. The conditions of the bearing surface in Figure 12 are a significant problem for a bearing that is expected to have reliable life over several years with only minimal lubrication. The surface damage was caused by wear debris generated from the pocket and likely aggravated by impact forces from ball collisions. The increase in operating temperature is also probably related to this same surface degradation of the bearing steel. This would disrupt the EHD film and likely result in higher friction. The early carbon-phenolic cages were hand wrapped by Allcomp. Since that time, Allcomp has added a wrapping machine that uses uniform tension in the wrapping process. This will like reduce wear generation by producing a more uniform matrix with fewer voids. Another solution that AFRL/PRTM is pursuing is coating carbon-phenolic to enhance the wear resistance and lubricity. The early results with this approach look very promising.

5. Conclusions

In this effort, we examined carbon-phenolic as a potential replacement material for cotton-phenolic cages. In Part I, it was shown that the carbon-phenolic material has superior mechanical and thermal properties to cotton-phenolic. However, the bearings tested in Part II showed that the full-scale bearings encountered surface damage and higher operating temperatures than bearings fitted with cotton-phenolic cages. Both of these problems are attributed to wear debris released from the ball pockets during operation. There are potential ways to eliminate this wear debris. There are also substantial benefits to replacing a cotton-phenolic cage with a more advanced composite material cage. However, more research is required to find what that replacement material should be. If a carbon-phenolic material can be developed that does not generate wear debris, it will be superior to cotton-phenolic because of the material properties described in Part I.

6. References

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